

# A miniature thermoacoustic stirling engine

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## Abstract

A miniature thermoacoustic stirling engine was simulated and designed, having overall size of length 0.65 m and height of 0.22 m. The acoustic field generated in this miniature system has been described and analyzed. Some efforts had been paid to coupling and matching, and a miniature thermoacoustic engine and some extra experimental components have been constructed. Analysis and experimental results showed that to obtain better performance of the engine, the diameter of the resonance tube must be chosen appropriately according to the looped tube dimension and the input heating power. It provided an effective way to miniaturize the thermoacoustic stirling heat engine. The experimental results showed that the engine had low onset temperature and high pressure amplitude and ratio. With the filling helium gas of 2 MPa and heating power of 637 W, the maximal peak to peak pressure amplitude and pressure ratio reached 2.2 bar and 1.116, respectively, which was able to drive a refrigerator, a heat pump or a linear electrical generator. The operating frequency of the engine was steady at 282 Hz.

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**Keywords:** Thermoacoustic stirling engine; High frequency; Pressure ratio

## 1. Introduction

Based on ideally reversible heat transfer, traveling wave thermoacoustic engines can achieve higher efficiency than the standing wave thermoacoustic engines based on intrinsically irreversible heat transfer. A traveling wave thermoacoustic prime mover was originally proposed by Ceperley in 1979 [1,2]. In 1999, a new type of traveling wave thermoacoustic engine, which employed the inherently efficient stirling cycle, called thermoacoustic stirling heat engine (TASHE), was constructed and studied by Backhaus and Swift [3,4], who upgraded the thermal efficiency up to 30% and to a pressure ratio [5] of around 1.2 by introducing a 1/4 wave length resonator in a looped tube. The efficiency had reached a level comparable to that of a car engine. With high thermal efficiency and pressure amplitude, those engines could

be applied in liquefying natural gas [6] or driving a refrigerator [7] etc. The above mentioned TASHEs were large engines with resonators about 10 m long at the resonant frequencies of about some tens of Hz. Because of using the normal inert gases as working material, a low energy density results for the thermoacoustic engine. Small scale thermoacoustic engines bring an insight to help solve the weak point. Because of working in a resonant mode, the thermoacoustic engine's main geometry scale, such as the length of the resonator, decides the operating frequency. Small thermoacoustic engines must operate at high frequency. From the viewpoint of the thermal cycle, the higher operating frequency means higher power density, which solves the weak point of a thermoacoustic engine, and a lot of practical applications of thermoacoustic devices could become realities. However, high frequency thermoacoustic engines also will face a series of difficulties to be solved before it could be used, for example, how to acquire high pressure amplitude and performance overcoming the viscous losses and irreversible losses induced by high frequency,

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how to solve more serious wave disturbing problems, what is different for modeling the design and so on. Until now, there are only a few researches on the TASHE concerning relatively high frequency. In 2001, Biwa et al. built a TASHE in which a traveling wave mode (273 Hz) and a standing wave mode (100 Hz) were observed [8]. We also constructed a TASHE with a high frequency mode (528 Hz) and a low frequency mode (76 Hz) [9]. They all concentrated on the thermoacoustic mechanism, and the largest peak to peak pressure amplitude observed was below 0.29 bar. Obviously, such small pressure amplitude produced by the present small scale engines cannot be applied as practical devices.

This paper aimed at investigating a miniature TASHE to find a way to improve the pressure amplitude and ratio of the system, and actually, to realize the miniaturization of the TASHE. According to this aim, a miniature thermoacoustic stirling heat engine with a resonance tube only about 350 mm long has been investigated, constructed and tested.

## 2. Simulation method

A thermoacoustic engine with a looped tube and a resonator, shown in Fig. 1, is verified as a high efficiency stirling cycle, the type of which is chosen to investigate small scale

from the top of the main cold heat exchanger, going down through the regenerator and thermal buffer tube to the T type junction (now, for the positive coordinate,  $X = 0.185$  m is obtained) and then turning right through the resonator to the end of the cavity, where  $X = 0.776$  m. Here, the working gas was helium; the heating power and working pressure were 200 W and 2 MPa, respectively; and the regenerator was composed of 150 mesh stainless steel screen. In such system configuration, the overall size of the investigated thermoacoustic engine was around  $0.6 \times 0.25$  m. Thus, the planned operating frequency was about 400 Hz according to the traditional estimating method for operating frequency when we treat this engine as a  $\lambda/4$  system, the same as a large scale system [10,11].

According to linear thermoacoustic theory, the momentum, continuity and energy equations are as follows [12]:

$$\frac{dp_1}{dx} = -\frac{i\omega\rho_m}{A_{\text{fluid}}(1-f_v)}U_1 \quad (1)$$

$$\frac{dU_1}{dx} = -\frac{i\omega A_{\text{fluid}}}{\rho_m a^2}\left(1 + \frac{(\gamma-1)f_k}{1+\varepsilon_s}\right)p_1 + \frac{\beta(f_k-f_v)}{(1-f_v)(1-\sigma)(1+\varepsilon_s)}\frac{dT_m}{dx}U_1 \quad (2)$$

$$\frac{dT_m}{dx} = \frac{\dot{H} - \frac{1}{2}\text{Re}\left[p_1\tilde{U}_1\left(1 - \frac{T_m\beta(f_k-f_v)}{(1+\varepsilon_s)(1+\sigma)(1-f_v)}\right)\right]}{\frac{\rho_m c_p |U_1|^2}{2\omega A_{\text{fluid}}(1-\sigma)|1-f_v|^2}\text{Im}\left[\tilde{f}_v + \frac{(f_k-f_v)(1+\varepsilon_s f_v/f_k)}{(1+\varepsilon_s)(1+\sigma)}\right] - A_{\text{fluid}}K - A_{\text{solid}}K_{\text{solid}}} \quad (3)$$

thermoacoustic engines. For the purpose of miniaturization, in our calculations, the dimension of the looped tube were initialized; the length and inner diameter of the resonance tube were fixed as 0.35 m and 18–22 mm, respectively.

An  $X$  coordinate is set up for<sup>1</sup> convenience of calculations and analyses in this thermoacoustic engine. Here, the top of the main cold heat exchanger is set to be the origin of the coordinate ( $X = 0$ ). The negative direction of  $X$  is taken anticlockwise from the top of the main cold heat exchanger, through the compliance tube and inertance tube, to the T type junction, where we can obtain  $X = -0.415$  m. The positive direction of  $X$  also starts

The calculation for the small thermoacoustic stirling engine was from the origin along the established coordinate to each segment in terms of Eqs. (1)–(3), with pressures and volumetric velocities matched at the junctions between segments, where  $p_1$  and  $U_1$  are the pressure and volumetric velocity amplitudes, respectively,  $\omega$  is the angular frequency,  $\rho_m$ ,  $T_m$ ,  $c_p$ ,  $\gamma$ ,  $K$ ,  $\sigma$  and  $\varepsilon_s$  are the mean density, temperature, isobaric specific heat, specific heat ratio, thermal conductivity, Prandtl number of working fluid and factor of material properties, respectively, and  $f_k$  and  $f_v$  are the viscous and thermal functions, respectively,  $A$  is the flow area of the channel,  $A_{\text{fluid}}$  and  $K_{\text{fluid}}$  are the cross section area and thermal conductivity of the solid forming the channel, respectively,  $\dot{H}$  is the total power,  $i$  is the imaginary unit and  $\text{Re}$ ,  $\text{Im}$  and superscript  $\sim$  are the real part, the imaginary part and the conjugation of a complex quantity, respectively.

From previous research, it has been perceived that the stirling thermoacoustic engine was a coupled system between the looped tube and the resonator and its operating frequency was influenced strongly by the coupling grade. When the size of the system became smaller, the operating frequency would become more sensitive, and

<sup>1</sup> In fact, the model starts near the top of Fig. 1, just above the main cold heat exchanger. It integrates up, left, and down, through the compliance and inertance to the junction. It then starts again just above the main cold heat exchanger, going down through the regenerator and thermal buffer tube to reach the junction a second time, where complex  $p_1$  and  $U_1$  must match. The integration then proceeds through the resonator. A shooting method is used in the integrations. In the simulation, for convenience of calculations and analyses, we set up an  $X$  coordinate, and chose the top of the main cold heat exchanger as the origin. The coordinate values in fact stand for the dimensions of the system.

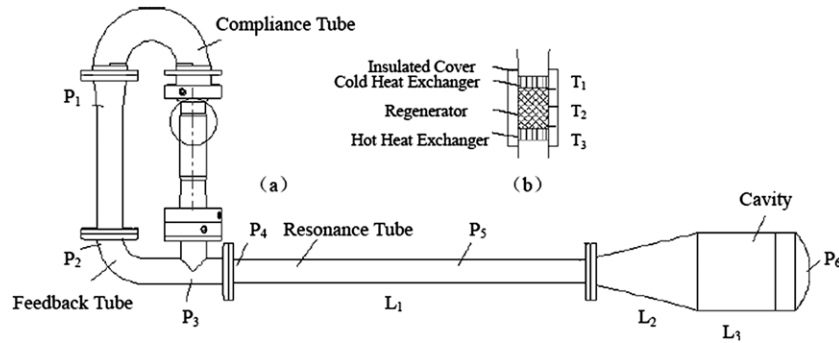


Fig. 1. Schematic of travelling wave thermoacoustic engine.

the acoustic field could also be deformed strongly. In the following, a series of calculations were considered and discussed.

### 3. The distribution of the acoustic field

Fig. 2 shows the computed pressure amplitude, velocity amplitude and the phase lead of the velocity relative to the pressure distribution of the miniature system, with the input power of 200 W and the filling helium of 2 MPa. Fig. 2a shows the axial distribution of the pressure amplitude  $P$ . The pressure amplitude is the maximum at the top of the cold heat exchanger and then decreases sharply in the regenerator. At  $X = 0.431$  m, the pressure amplitude becomes 0, which is the node of the pressure amplitude and is located at  $P_5$  in the resonator. After that, the pressure amplitude begins to increase gradually to a new extremum at the end of the cavity, which can be regarded as a new pressure antinode, and it is smaller than the maximum pressure amplitude. Obviously, this system is neither  $\lambda/2$  nor  $\lambda/4$  exactly, so the miniature thermoacoustic stirling heat engine can not be simply treated as a  $\lambda/2$  or  $\lambda/4$  system.

The velocity amplitude distribution of the system is shown in Fig. 2b. The velocity minimum in the looped tube of the system is near the location of  $X = 0.015$  m, which is right at the top of the cold end of the regenerator. The velocity amplitude approaching 0 near the regenerator will contribute to reducing the viscous loss in the regenerator, which is helpful for improving the efficiency of the regenerator and the performance of the TASHE. In addition, it can be seen that there is an abrupt point of  $X = 0.184$  m in Fig. 2b; in fact, the abrupt point of  $X = 0.184$  m is the T type junction, where the sum of the velocity of the feedback and the velocity of the bottom of the second cold heat exchanger will make the velocity at the beginning of the resonator. So, the velocity in the resonator is not continuous and could be larger than the velocity in the looped tube, which means much acoustic power will be dissipated in the resonator, and it may dominate the main viscous loss of the system under some conditions.

Fig. 2c presents the distribution of the lead phase of the velocity relative to the pressure. The phase lead of the

velocity relative to the pressure decreases gradually from the T type joint ( $X = -0.415$  m), the position near the regenerator and becomes a pure travelling wave phase at  $X = 0.009$  m, close to the cold end of the regenerator. On the other hand, from Fig. 2c, in the resonator, the lead phase of the velocity relative to the pressure is  $-90^\circ$  first, and after the pressure node ( $X = 0.436$  m), it becomes  $90^\circ$  gradually. Obviously, according to the relation of the phase difference, it seems to be a  $1/2$  wave length standing wave in the resonator, which should determine the operating frequency of the whole system, but in fact, the frequency can not be determined by this factor alone. From our previous analysis, the concept of impedance has been applied to the analysis [12]. The calculated operating frequency was about 291 Hz here, which greatly deviates from the frequency of a  $1/2$  wave length system that is predicted at 700 Hz! It reveals that the length of the resonator would not be the only effective factor to determine the resonant frequency in the miniature thermoacoustic engine.

As it is found that the pressure amplitude and velocity amplitude take maximum and minimum values close to the cold end of the regenerator, respectively, so, the acoustic impedance  $Z$  would reach the maximum near the regenerator, and therefore, the viscous losses in the regenerator can be well suppressed. On the other hand, the miniature thermoacoustic stirling engine can not simply be treated as a  $\lambda/2$  or  $\lambda/4$  wave length system. It is decided by this coupling system of the looped tube and the resonator, and the frequency should be determined by the whole system, such as the length and inner radii of the resonator tube, feedback tube and compliance tube and so on. In large scale systems, the inner radii of tubes have few influences on the working frequency and the performance; and therefore, they always can be neglected. In small scale systems, from the analysis of the acoustic field in the thermoacoustic engine, it is obvious that they can not be ignored. In the following calculations and experiments, we will reveal this.

### 4. Experimental apparatus and verification

Based on the calculations above, a miniature thermoacoustic stirling engine was constructed, shown in Fig. 1. The engine consisted of two parts, the traveling wave

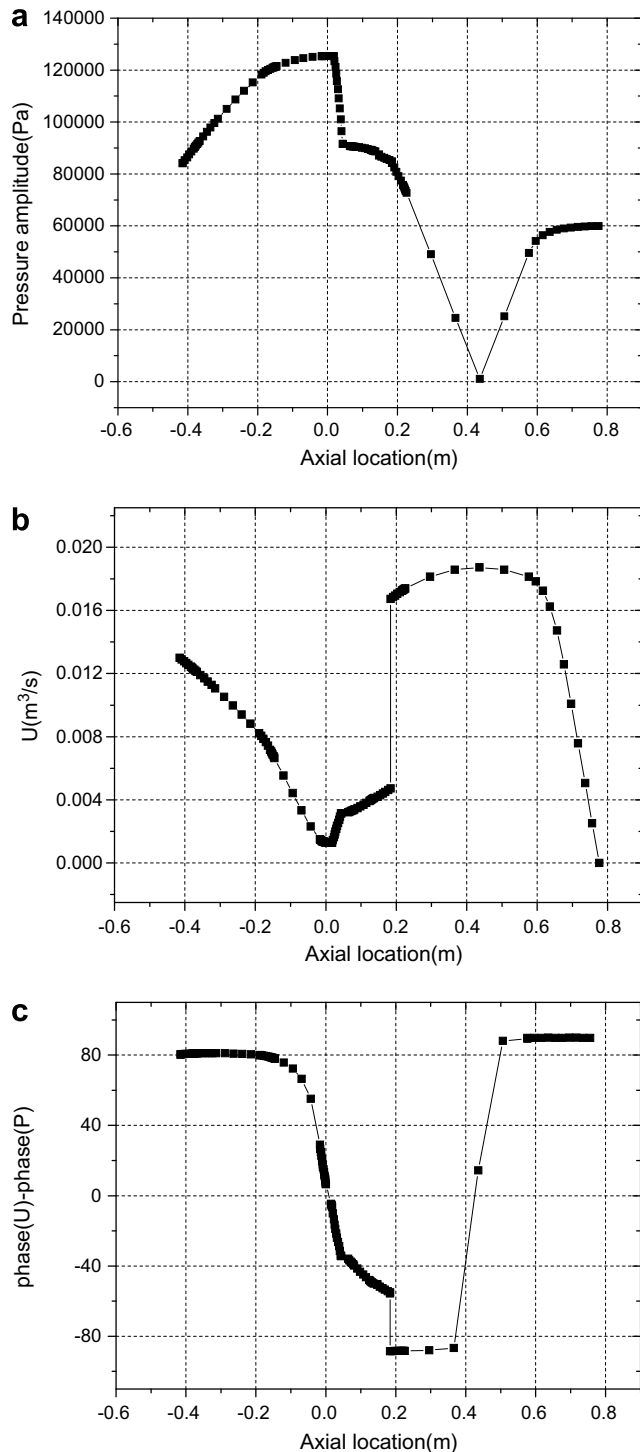


Fig. 2. The computed pressure amplitude, velocity amplitude and the phase lead of the velocity relative to the pressure distributions of the miniature system.

looped tube and the standing wave resonator including a straight pipe and a cavity, which were of 20 mm and 70 mm inner diameter. The total length of the looped tube was about 0.6 m and that of the resonator L1 was about 0.35 m; both L2 and L3 were 0.1 m. The regenerator was 25 mm long and of 150 mesh stainless steel screens. The calculated volume porosity was about 0.699. Near the

top of the regenerator, there was the main cold heat exchanger, which was a parallel plate pattern copper heat exchanger.

In the experiments, the pressure data are acquired and analyzed with no time delay by a computer. As shown in Fig. 1, pressure sensors labeled  $P_1$ ,  $P_2$  and  $P_3$  are located at the end of the compliance tube, inertance tube and the T type junction, respectively;  $P_4$ ,  $P_5$  and  $P_6$  are located along the resonance tubes, in order to observe the distribution of the acoustic field in the engine. The pressure data acquisition system is comprised of the pressure sensors, data acquisition board, PIII computer and self developed program based on VB program language. The pressure sensors are of the piezoelectric type (CY-YD 208), supplied by Sino-cera Piezotronics, Inc. China. The data acquisition board is AC1810, 12 bits precision and 10 ks/s sampling frequency. The output voltage from the piezoelectric transducer is amplified by a charge amplifier and a signal amplifier, and sent to the AC1810 board in the computer. Through execution of the self developed program, the pressure signal can be acquired in the screen of the computer. Point  $P_1$  is taken as the representative checking point for it is the biggest pressure amplitude point among the six positions and is, therefore, important for connecting the refrigerator or other applications. The phase of the pressure signal and operating frequency is measured by a Stanford Research model SR-830 DSP lock in amplifier. Also, three thermocouples are placed along the regenerator and are shown as  $T_1$ – $T_3$  in Fig. 1, which are measured by calibrated NiCr–NiSi thermocouples (with  $\pm 1$  °C accuracy). The temperature signal can be acquired by the computer through a modern digital multi-purpose meter and its driving software, provided by the UNI-T, Inc. Hong Kong. Heating power is adjusted by changing the charging voltage to the heaters and is displayed by a digital dynamometer.

#### 4.1. Onset process

The thermoacoustic stirling engine works in a resonance condition, the onset process is the starting process, which can bring forth the basic feature of the engine. Fig. 3 shows the time developing curve of the onset process at  $P_1$  in the looped tube. The operating conditions were mean pressure of 2.0 MPa, helium gas and heating power of 195 W. The temperature of the hot end of the regenerator at the  $T_3$  thermocouple increased gradually and did not generate any sound wave in the system at first. When the temperature difference across the regenerator increased to 418 °C, larger than the critical temperature difference, oscillation of the engine was generated. The looped tube had a comparatively high onset pressure ratio at very short time, and its value at  $P_1$  could even reach about 1.1. Thereafter, the pressure amplitude and also the temperature at  $T_2$  decreased sharply and maintained a steady state later. The steady heating temperature at the hot end of the regenerator was about 316 °C. That was because the heating power was converted into acoustic power at the moment

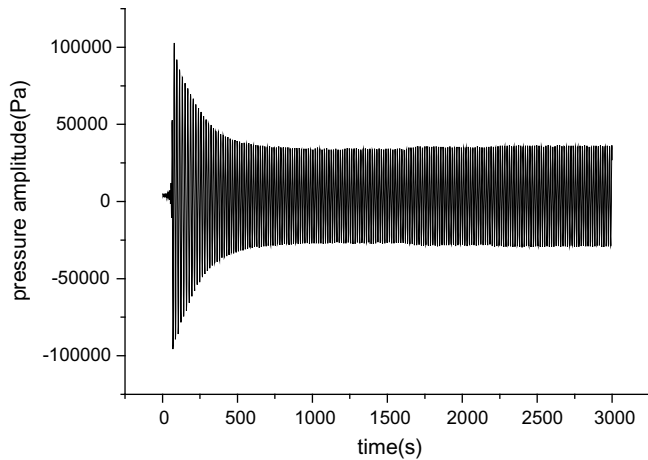


Fig. 3. The time developing curve of the onset process.

when the temperature difference exceeded the critical temperature difference, and then the acoustic wave propagated through the whole system, eventually reaching a steady pressure amplitude.

According to the experimental results, it can be concluded that the engine can self oscillate steadily without higher harmonics, and the onset temperature is also low, which means that the basic performance of the constructed engine is rather good and is a steady system.

#### 4.2. Pressure waves

Fig. 4 shows typical pressure waves at the different locations when the engine operated steadily with the mean pressure of 2 MPa and heating power of 560 W. The maximal peak peak pressure amplitude and pressure ratio reaches 0.202 MPa and 1.107 at  $P_1$ . It can be found that the pressure wave is a good sinusoidal pattern with very small phase differences at  $P_1$ ,  $P_2$ ,  $P_3$  and  $P_4$ . The phase difference between  $P_4$  and  $P_6$  is about 180 deg, which means that the

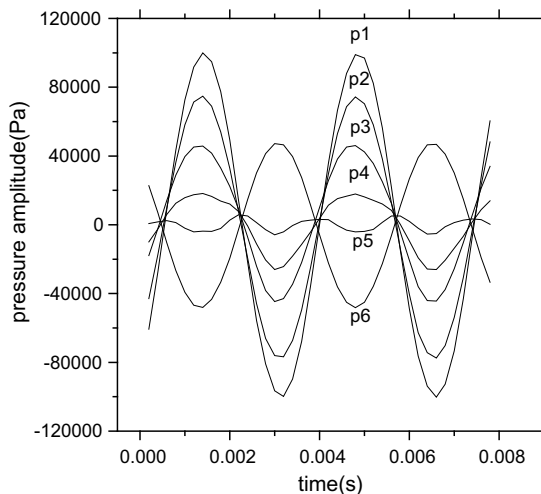


Fig. 4. Typical pressure waves.

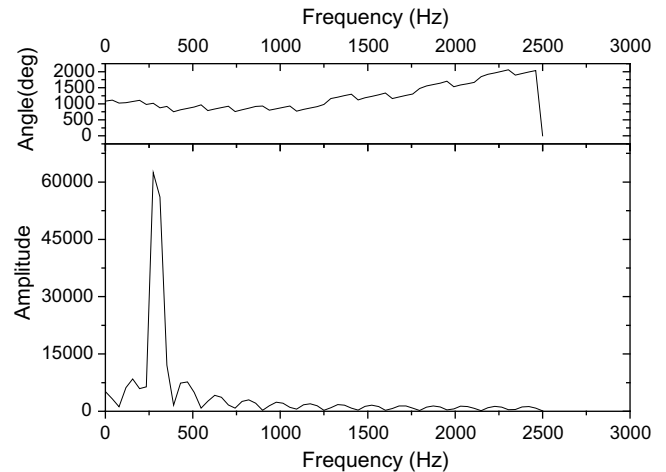


Fig. 5. Relative Fourier component at  $P_1$ .

wave in the resonator is not a  $1/4$  wavelength, instead, it agrees well with the calculated results above. The amplitude at  $P_5$  is much smaller, and there is no good sinusoidal wave because  $P_5$  here is designed as the pressure node of the standing wave. From Fig. 4, it can also be seen that the period of the system is about 3.54 ms, which means that the operating frequency of the system is about 282 Hz. It shows a great discrepancy with the estimated frequency of 400 Hz according to the traditional method. The relative Fourier component at  $P_1$  is shown in Fig. 5. It's obvious that the present system oscillated stably at the resonant frequency of 282 Hz. In our experiments, no higher frequency modes have been observed as in the literature [8,9], which meant that the looped tube and resonator were well matched.

#### 4.3. Optimization of the resonance tube diameter

Analysis and simulations of the acoustic field above have revealed that in the small scale stirling thermoacoustic heat engine, the length of the looped tube was comparable to that of the resonance tube; and therefore, the length of the resonance tube was not the dominant factor to determine the operating frequency and the performance of the system any more. The radial dimension of the resonance tube might play an important role in this miniature system. Now, it is necessary to verify this through experiments and calculations. Experiments were conducted with helium as the working gas, 2 MPa mean pressure and 15–22 mm inner diameters of the resonance tube.

The relation of the measured operating frequency versus the resonance tube diameter is shown in the Fig. 6. It can be seen that an increase of the resonance tube diameter leads to a remarkable increase of the working frequency. According to the figure, when the resonance tube diameter increases from 15 mm to 22 mm, the working frequency increases from 226 Hz to 304 Hz, by 34.5%. The computed results and the estimated operating frequencies by Qiu [11]



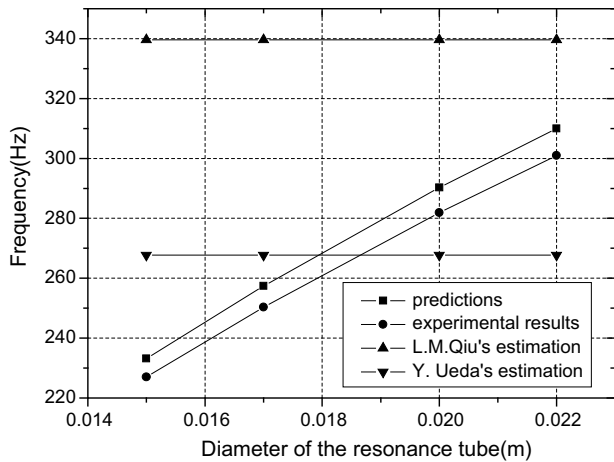


Fig. 6. Working frequency vs. diameter of the resonance tube.

and Ueda [13] are also presented in Fig. 6. The computed curve is almost consistent with the experimental data, and the errors are less than 5%. However, Qiu's and Ueda's estimated results have great errors relative to the experimental data. In this case, a thinner resonance tube means lower resonant frequency, which is helpful for improvement of the performance of thermoacoustic engines and the frequency match between the thermoacoustic engine and a pulse tube refrigerator or electrical generator. In this way, also, it can be concluded that if we keep the frequency constant, we have to shorten the length of the resonance tube, which is significant for miniaturization of a traveling wave thermoacoustic engine.

The remarkable changing of the operating frequency inevitably brings about the changing of the performance of the system. To observe the relationship between the pressure amplitudes and the diameter of the resonance tube, the heating power is fixed to a certain value. Point  $P_1$  is taken as the representative checking point for it is the highest pressure amplitude point among the six positions and is, therefore, important for connecting the refrigerator or other

applications. Fig. 7 shows the comparisons of the peak to peak pressure amplitudes between the experiments and calculations. It can be seen that with the diameter of the resonance tube decreasing, the pressure amplitude increases first, but when the diameter is further reduced, the pressure amplitude begins to decrease. In addition, in the figures, we found that with the heating power increasing, the optimal diameter to obtain maximum pressure amplitude is different. The trends of the experimental results are similar to the computed ones. In our experiments, when the heating power is below 495 W, the optimal diameter is 17 mm and the maximum peak to peak pressure amplitude is 0.184 MPa, but when it achieves 560 W, the optimal diameter becomes 20 mm, and the maximum peak to peak pressure amplitude is 0.2 MPa. So, if the input heating power increases, we should choose a larger resonance tube to obtain the maximum pressure amplitude.

However, in our experiments, the heating power is 3–4 times the amount of the computed one to reach the same pressure amplitude and pressure ratio. In fact, there are many possible heat loss mechanisms that were not taken into account in the simulations. Firstly, although thermal insulation is provided for the components at the high temperature regions in the experiments, heat leakage to the surroundings is unavoidable. Secondly, the simulation model has not considered the reflection loss and turbulence loss induced by the changing the cross section, which could be the main losses in an acoustic system. Thirdly, the Gedeon streaming, a time averaged mass flow associated with acoustic work flow, which can transport heated gas from the hot heat exchanger through the annular space to the ambient exchanger has not been suppressed effectively. Also, since this is the first high frequency miniature TASHE, the regenerator and some structures of the system have not been optimized. Therefore, the actual heat reacting on the thermoacoustic process was only a part of the total input heat power. In our simulations, we only chose the actual heat as the input power. In future work, we will

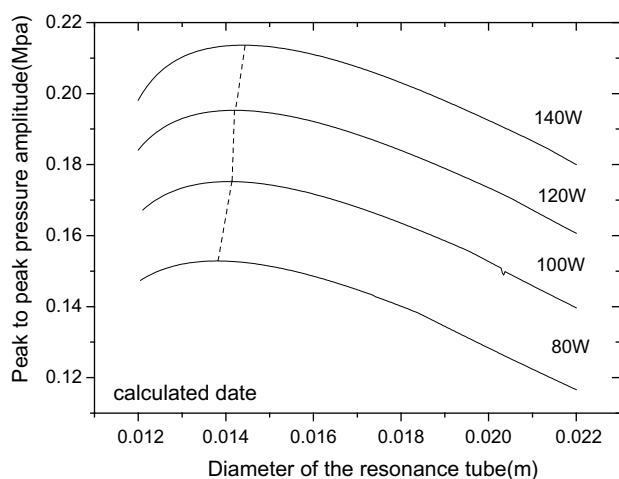
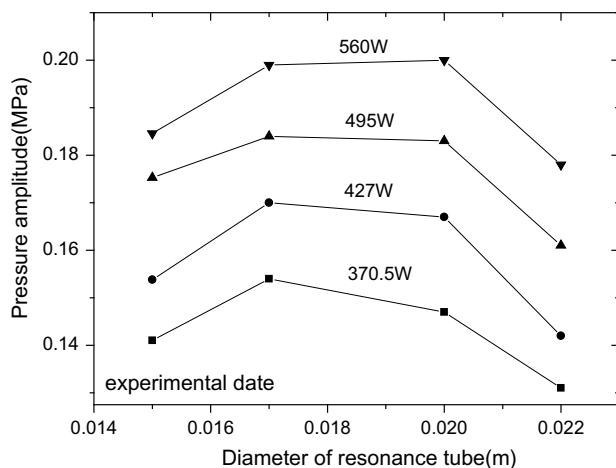


Fig. 7. Pressure amplitude vs. diameter of the resonance tube at certain input power.

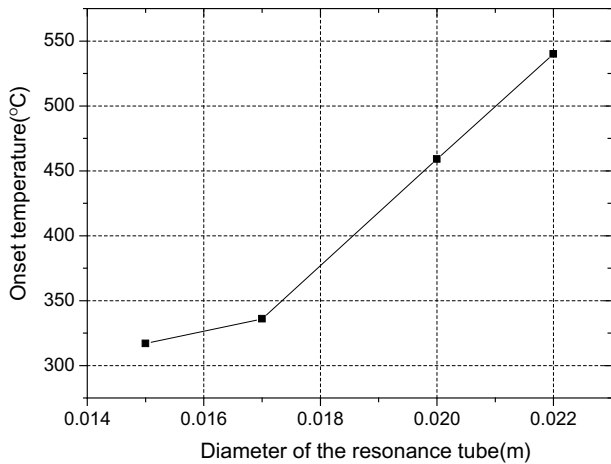


Fig. 8. Onset temperature vs. the resonance tube diameter in the experiments.

contribute our attentions on solving those problems in our simulation program.

Fig. 8 shows the relation of the onset temperature and the resonance tube diameter. It is clear that the onset temperature increases with the resonance tube diameter increasing because of the operating frequency increasing. So, the thin resonance tube is easier to oscillate, which is significant for utilization of low grade energy.

Fig. 9 shows the comparisons of the heating temperature between the experiments and the calculations, while the input power is fixed to a certain value. It is obvious that the trends of the curves are in agreement with the computed curves. From the figures, although the onset temperature is relative low, the thin resonance tube's operating temperature increases sharply with increasing input heating power. The thin resonance tube should not be used in large scale systems in which the heating power is huge. For big diameter resonance tubes, the heating temperature relative to the input heat is not sensitive. In the figure, if the diameter is 20 mm, when the heating power increases from 370.5 W to 560 W, the heating temperature increment is only 15.5 K, whereas,

if the diameter is 15 mm, the increment is 47.5 K. Also in our experiments, in the figure, when the resonance tube diameter is 22 mm, the temperature difference between the maximum and minimum is only about 5 K, but here, increasing the heating power actually brings about a decreasing of the heating temperature. The reason is still under research. So, the larger resonance tube can receive more heat. In our experiments, even when the input heat achieved 560 W, the heating temperature was only 307.5 °C.

We aimed to improve the pressure amplitude in the thermoacoustic engine as high as possible, which could be used to drive pulse tube refrigerators and electrical generators. More heating power meant higher pressure amplitude. In our design, according to the simulations and experiments, the diameter of the resonance tube was chosen as 20 mm. In this case, until now, the maximal heating power achieved in our engine was 637 W and the maximal peak to peak pressure amplitude reached 0.22 MPa.

## 5. Conclusions

A miniature thermoacoustic stirling engine has been calculated and realized. This engine was operated in a traveling wave mode in the looped tube and a hybrid mode of standing wave in the resonator tube. The simulation and experiments revealed that for a miniature thermoacoustic engine, the acoustic field generated in the system could not be considered as a simple  $\lambda/2$  or  $\lambda/4$  standing wave field, which meant that the operating frequency could not be estimated simply by the resonator tube length, whereas for a large scale thermoacoustic stirling heat engine, this estimating method for operating frequency had been used for several years widely [10,11].

Experiments showed that an appropriate resonance tube diameter was very important to decrease the operating frequency and improve the pressure amplitude if the longitudinal dimension had been fixed, which is crucial in the miniaturization of the TASHE. When the resonance tube diameters were reduced from 22 mm to 15 mm, the

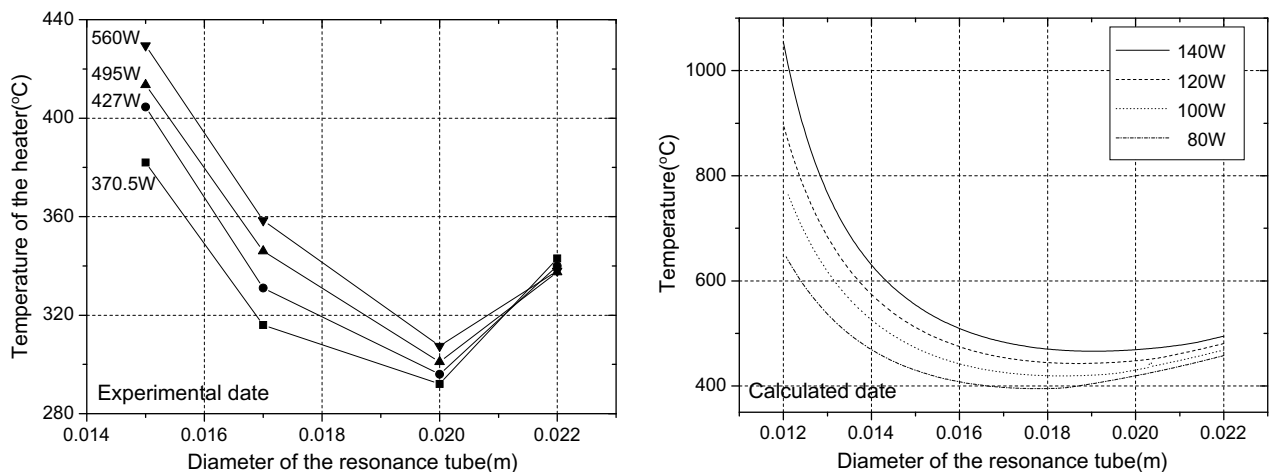


Fig. 9. Heating temperature vs. diameter of the resonance tube.

working frequency decreased from 226 Hz to 304 Hz, by 34.5%. In our experiments, with the resonance tube length of 350 mm, a 20 mm diameter resonance tube was chosen as the best, and the operating frequency was about 282 Hz. With 2 MPa helium gas as working gas, 638 W of heating power, the maximal peak to peak pressure amplitude reached 0.22 MPa and the heating temperature was only 310 °C, which was able to be applied in the thermoacoustic refrigerator or other applications.

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